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Loop Heat Pipe Operating Temperature Dependence on Liquid Line Return Temperature

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ABSTRACT

A Loop Heat Pipe (LHP) is a passive two-phase heat transfer device developed and successfully employed to cool spacecraft (satellite) electronics. The intrinsic benefits of this technology (lightweight, small volume, high thermal conductance) make it an attractive potential solution to many problems in ground vehicle thermal management. As most published LHP research has focused on cooling orbiting spacecraft components, there is little knowledge of how LHPs perform under the temperature extremes (-40°C to 40°C) and diurnal/seasonal fluctuations anticipated with terrestrial applications. Ambient temperature extremes mandate consideration of transport line heat exchange with the surroundings (parasitic losses/gains).

This paper presents results from an experimental investigation of liquid line return temperature impact on system performance for sink temperatures from –30°C to 40°C and evaporator loads up to 700 Watts. A heat exchanger is placed on the liquid line to control the temperature of the fluid returning to the compensation chamber and evaporator, and to compensate for heat exchange between the subcooled liquid and the environment. Control of the returning liquid temperature enables simulation of any subcooling condition desired, and evaluation of the impact on system performance from heating or cooling the liquid line.

INTRODUCTION

Adapting Loop Heat Pipe technology to terrestrial applications (e.g. cooling of ground vehicle systems) poses significant technical questions, particularly with regard to heat sink and ambient dynamic range. Sink temperatures for ground vehicle application vary from -40 to 40°C. This may compromise system performance by driving the loop operating temperature to unacceptable levels to satisfy condenser heat rejection requirements. Increasing condenser size to minimize the condenser sink temperature difference is generally unacceptable from a vehicle integration standpoint. Therefore, LHP design optimization is critical. The sizing of a LHP for a vehicle application requires an understanding of how the operating temperature varies with the liquid line inlet temperature and applied heat load. In addition the proper evaporator must be chosen based on system design and the environment.

This experiment is an investigation of liquid line return temperature impact on system performance for temperatures from –30°C to 40°C and evaporator loads up to 700 Watts. Heating and cooling blocks are placed on the liquid line to control the temperature of the fluid returning to the evaporator/compensation chamber and to compensate for heat exchange between the subcooled liquid and the environment. Control of the returning liquid temperature enables simulation of any subcooling condition desired, and characterization of system performance.

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BACKGROUND

The temperature of the compensation chamber directly affects the loop operating temperature. Any factors that affect the compensation chamber energy balance impacts the entire loop operating temperature profile. The compensation chamber can exchange energy with the environment, the evaporator, and the liquid returning from the condenser. Heat transfer with the environment can sometimes be ignored for a well-insulated compensation chamber. In this case the energy balance in the compensation chamber is a function of the subcooled liquid and the heat leak from the evaporator.

EVAPORATOR HEAT LEAK - The heat transfer between the compensation chamber and the evaporator depends on the fluid quality in the evaporator core [1]. If the evaporator core is completely filled with liquid (quality of zero), heat transfer is achieved mainly by conduction through the hermetic case and is relatively small. If the evaporator core contains vapor then thermodynamically it becomes a heat pipe transferring energy from the evaporator core to the compensation chamber. Heat transfer occurs through the primary wick and is transmitted to the compensation chamber by evaporation and condensation along the vapor arteries. Heat transfer under this mode can be orders of magnitude higher than by heat conduction through the hermetic case alone. The heat transmitted from the evaporator to the compensation chamber is commonly referred to as the heat leak.

RETURN FLUID MASS FLOW - The heat exchange between the compensation chamber and the returning liquid depends on the mass flow rate and the temperature of the returning liquid. Mass flow is a function of the evaporator load (minus the evaporator wick heat leak) and the fluid latent heat of vaporization at the operating temperature. The working fluid mass flow rate is proportional to the ratio of thermal load and the latent heat (eq. 1),

$$\dot{m} = \frac{Q_{load} - Q_{heatleak}}{h_{fg}} \quad (1)$$

where the latent heat is based on the saturation temperature in the compensation chamber/evaporator. Increases in the evaporator load result in increased mass flow. A changing saturation temperature affects the fluid properties including the latent heat (e.g. increased temperature reduces the latent heat of vaporization), however, the change in latent heat is relatively small over the temperature range of interest.

LIQUID LINE RETURN FLUID TEMPERATURE - Return fluid temperature is a critical aspect of the compensation chamber energy balance and depends on system design and environmental variables. Heat exchange along the transport lines dominate return fluid temperatures in many scenarios. The effects of transport line heat exchange decrease substantially for cases of higher mass flow and lower sink to operating temperature gradients.

TEST ARTICLE AND SET-UP

Figure 1 shows a schematic and accompanying photo of the test article. The evaporator and the compensation chamber CC are stainless steel with outer diameter of 25.4 mm, and lengths of 30.5 mm (evaporator) and 12.7 mm (CC). The primary wick is made of sintered powder nickel with a pore size of 1.3 microns and a permeability of 1.3×10^{-14} m^2. The vapor line has an I.D. of 3.34 mm and a length of 1.94m while the liquid line has an I.D. of 1.75 mm and a length of 2.1m. The condenser line has an I.D. of 3.86mm and a length of 1.99m. Ammonia is used as the working fluid. An aluminum block with two cartridge heaters is attached to the evaporator to provide a total power up to 1200W. A refrigerator cools the condenser. There are a total of 39 thermocouples that monitor the loop temperatures. A differential pressure transducer is installed across the evaporator and CC to measure the external pressure drop. A data acquisition system and the LabView software programs are used to monitor and store data. The simultaneous collection and display of temperature and differential pressure data using two computers and two monitors proves to be crucial in the understanding of interactions among various elements in LHP operation.

A series of tests, at various evaporator loads (100, 300, 500, 700W) and liquid return temperatures varying from –30°C to 40°C provide insight into the evaporator/CC thermal response. The evaporator/CC inlet fluid temperature is controlled through the combined working of the condenser sink chiller, and liquid line heating element and/or cooling shell and tube heat exchanger. The condenser chiller provides a cold reference sink from which to cool or heat the liquid temperature accordingly. The condenser chiller minimum temperature is –27°C. The liquid line chiller minimum is –60°C. Actual inlet temperature minimums for the evaporator/CC inlet fluid temperature are limited by system inefficiency and losses. The minimum achieved based on the combined contributions is –30°C.

RESULTS

Test results suggest a linear correlation between the liquid line return temperature and the compensation chamber temperature for the given test conditions. Figure 2 shows compensation chamber temperature (average of TC1-TC5) as a function of inlet temperature for a constant 100W evaporator load. Linear models describe the thermal relationship. This linear response is consistent across all four evaporator loads, 100W, 300W, 500W, 700W, see figure 3.

The compensation chamber temperature is a function of evaporator heat leak, ambient exchange and liquid line

temperature, as described earlier. A linear model characterizes the response using a single variable, liquid line return temperature, while neglecting the others. A coefficient of determination, R^2, value provides some measure of the linear models overall performance. It is defined as the ratio of model sum of squares to the total sum of squares. A combined plot R^2 value is .9604, figure 3. The linear model for the combined plot is:

$$T_{cc} = 0.7178 * T_{LL\,Return} + 87.04 \tag{2}$$

The relatively large R^2 values imply that a linear model correlating the compensation chamber temperature to the liquid line return temperature accounts for 96.04% of the combined data variability. Other variables, such as ambient heat exchange with the CC or heat leak from the evaporator, account for only 4% variability between the measured data and the linear model. This suggests ambient heat exchange and evaporator heat leak play a secondary role in compensation chamber temperature for the given test conditions.

The average CC temperature (mean of TC1-TC5) increases with liquid line return temperature (Fig. 3). The relationship between liquid line temperature and CC is not affected significantly by evaporator load. This is not surprising since the additional heat exchangers on the liquid line drive the liquid line return temperature and negate any influence by the condenser or transport lines. Essentially we are setting conditions similar to an infinite condenser and adiabatic liquid line where system design and environment have little effect.

An interesting result, visible in figure 3, is the increasing compensation chamber temperature variance for liquid line return temperature below 268K. Data above 268K show reasonably small (<5°C) CC temperature spreads at a given liquid line return temperature. However the CC temperature spread increases below 268 K. This suggests that the ratio of evaporator heat leak to vapor producing evaporator load is changing. Consider an equation for the ratio of evaporator heat leak to the vapor producing evaporator load. Equation 3 is derived assuming ambient-CC heat exchange is negligible. Combining equation 1 with the CC energy balance and rearranging produces:

$$\frac{Q_{heatleak}}{Q_{evapload}} = \frac{C_p}{h_{fg}}(T_{cc} - T_{LL\,Return}) \tag{3}$$

At 273 K, the CC temperatures are extremely close. Therefore the ratio derived in equation 3 is equal for all four power settings. Below 268 K the ratio, Eq. 3, changes as a function of power. Thus the percent evaporator heat leak is greatest at 700 W.

Typically, increased evaporator loads place additional demands on the condenser ultimately causing increased loop temperatures after constant conductance is achieved. Heat exchange along the liquid line can also produce loop temperatures as described earlier. The V-shape response in figure 4 is a by-product of the various thermal contributions from the transport lines and condenser.

At a very low heat load, the condenser is not fully utilized and the liquid leaves the condenser at a temperature close to the sink temperature. Assuming the local liquid line ambient is higher than the sink temperature (positive heat exchange) the liquid is heated as it flows along the liquid line. For low mass flows its temperature will approach the ambient temperature as it enters the compensation chamber.

For moderate heat load increases a higher mass flow rate brings more subcooled liquid to the compensation chamber and the loop operating temperature decreases. This trend continues until the condenser is fully used. Further increases in the heat load require an increase in the operating temperature to dissipate the energy at the given sink temperature. This trend produces the v-shaped curve shown in figure 4. This suggests, for terrestrial applications, the corresponding system design and integration may be the dominant challenge for passive technologies.

Figure 5 demonstrates liquid line heating for two mass flows corresponding to 100W and 400W, respectively and a constant sink temperature of –20°C. The ambient laboratory temperature is 22°C. Figure 5 shows a 32°C temperature increase along the axial direction of the liquid line at 100W but only an 8°C temperature increase along the axial direction of the liquid line at 400W. The difference in temperature rise results from the increasing mass flow at higher applied powers. Increased mass flow rates result in lower sensible heating of the single-phase liquid because less time is available for heat transfer with the ambient.

RELEVANCE

A connection is possible between the fundamental evaporator/compensation chamber response in figure 3 and the V-shape response in figure 4. Analytically or experimentally derived liquid line return temperatures can be used with the fundamental response, Eq. 2, to determine the corresponding compensation chamber temperature. Table 1 shows experimental and analytical comparisons for a test where the liquid line return temperature was not controlled. The analytical results are calculated using the linear model for the combined data, equation 2.

A comparison of the measured and calculated results shows very good correlation over a fairly broad range of evaporator loads, figure 6. It should be noted that the liquid line return temperature data represent a fairly small segment on the response in figure 3. Results at lower temperatures, < 268K, may produce more significant differences between measured and calculated

CC temperatures due to the greater variation from the linear model.

Differences between the experiment and analytical data may result from experimental error and the 4% error between the linear model and the experimental data. Experimental error can result from measurement accuracy and experimental uncertainties. (i.e. evaporator core quality). Randomized experimentation and replication are means to characterize the experimental uncertainties.

Table 1: Comparison between Analytically or experimentally derived liquid line return temperatures

Evaporator Load	Liquid Line Return Temperature	Measured Compensation Chamber Temperature	Calculated Compensation Chamber Temperature based on linear model
100	287	295	293.1
200	278	285	286.6
400	276	283	285.2
600	286	292	292.3
600	276	282	285.2
400	271	278	281.6
400	277	283	285.9
200	275.5	283	284.8
200	277	284	285.9

CONCLUSION

A significant amount of literature discusses the V-shape curve depicted in figure 4 as a LHP characteristic response. [2, 3, 4] This does represent a characteristic response for a specific scenario. The curve shape is largely dependent on the system design and environment. The ideal LHP would transfer energy along adiabatic lines and use an infinite condenser providing negligible temperature difference with the sink. The fundamental response represents this optimum performance. System design and environmental effects cause deviation from the fundamental response and development of the response noted in figure 4. Robust system design and integration require knowledge of the fundamental temperature response of the evaporator/compensation chamber component.

Test results suggest a linear correlation between the liquid line return temperature and the compensation chamber temperature. This interdependency accounts for most of the data variability. Other variables, such as ambient heat exchange with the CC or heat leak from the evaporator, account for only 4% variability between the measured data and the linear model suggesting ambient heat exchange and evaporator heat leak have little to do, in this case, with compensation chamber temperature.

Analytically or experimentally derived liquid line return temperatures used with the fundamental response curve yield the corresponding compensation chamber temperature. This suggests design engineers, without specific LHP expertise, may possess the ability to conduct system integration and optimization for their particular application. Additional work is necessary to determine the robustness and tune-ability of the fundamental response slope.

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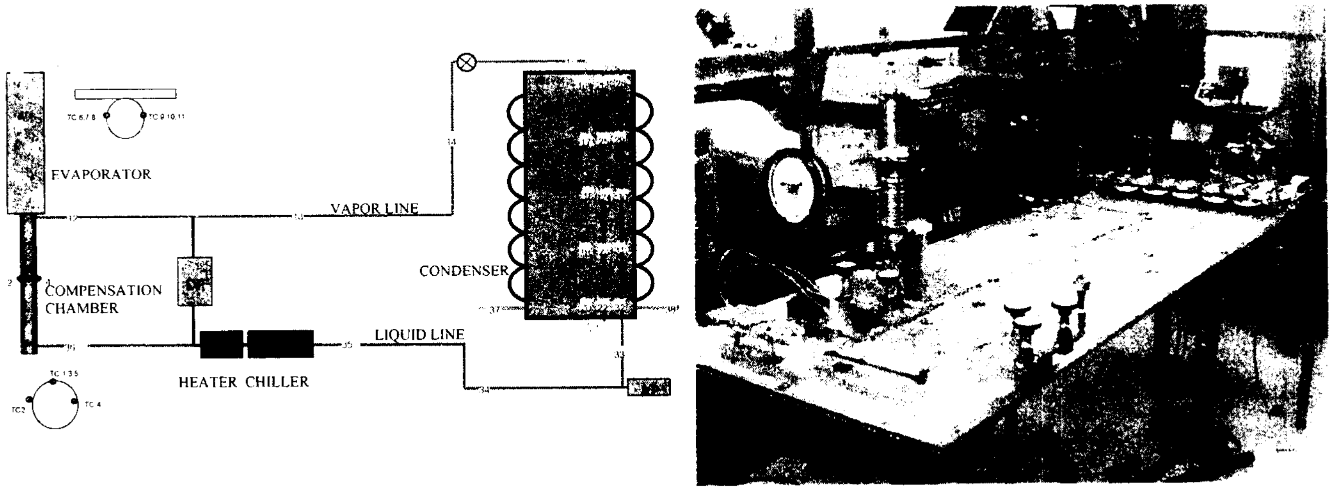

Figure 1: (Left) Test apparatus schematic showing thermocouple locations and pressure measurements. (Right) Photograph of test apparatus.

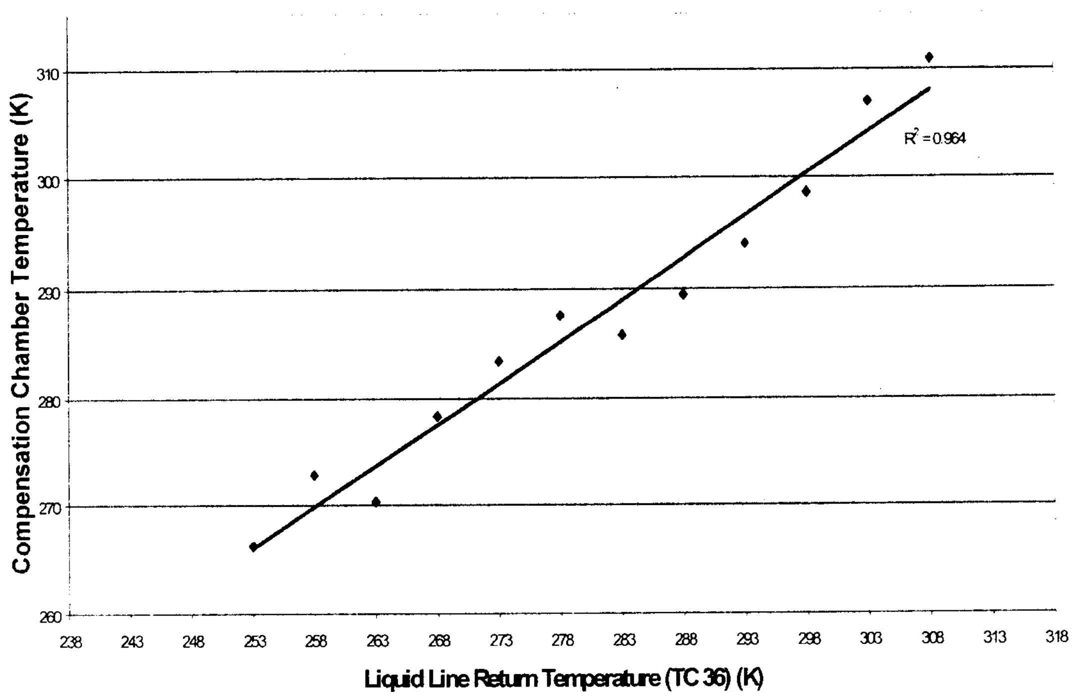

Figure 2: Thermal response for tests at 100W.

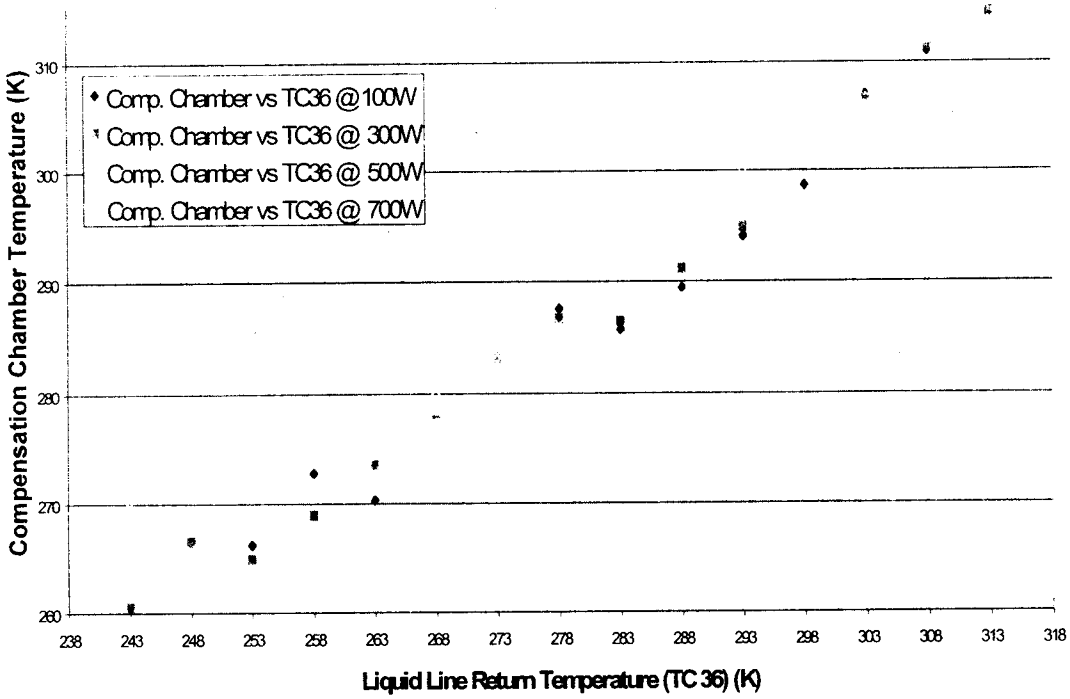

Figure 3: Thermal response curves for combined evaporator loads.

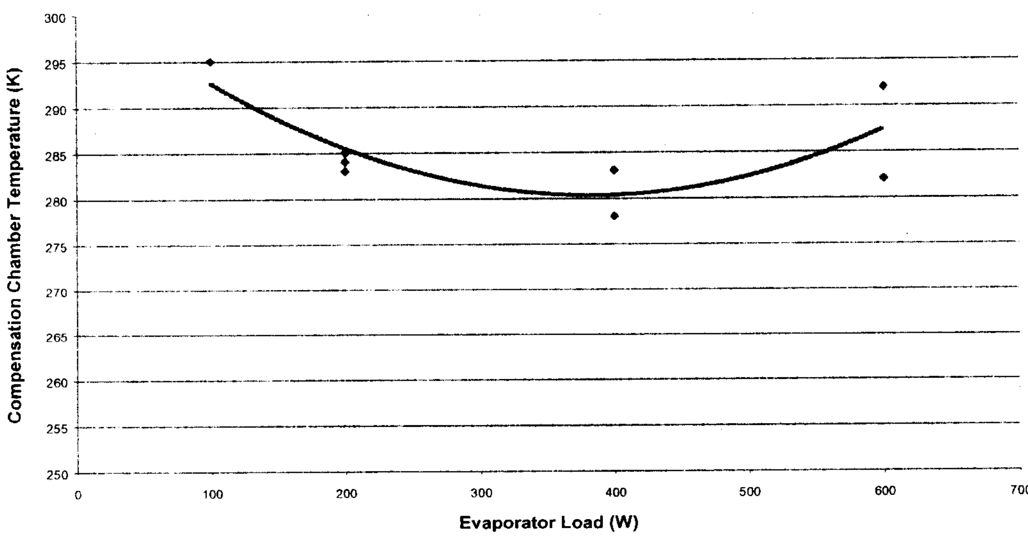

Figure 4: Characteristic compensation chamber response for conditions with the ambient temperature higher than the condenser sink.

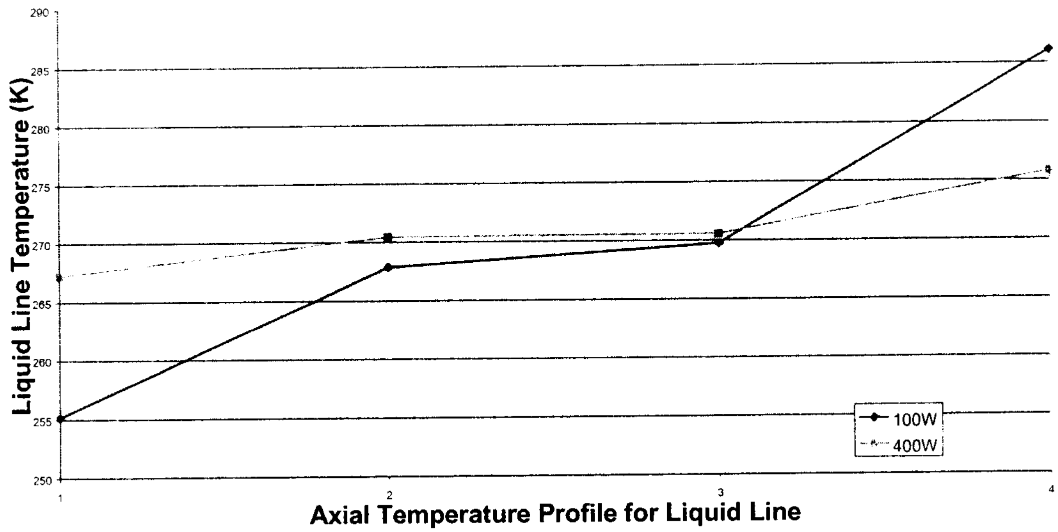

Figure 5: Data showing different liquid line temperature profiles as a function of power at -20°C Sink.

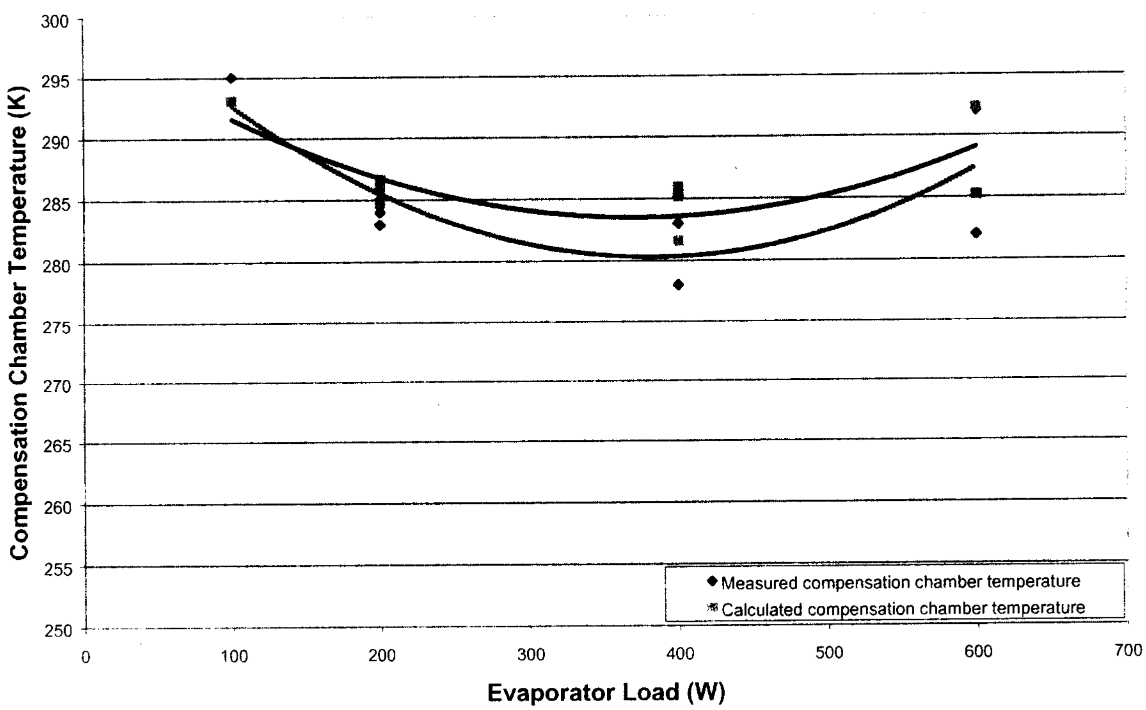

Figure 6: Experimental vs. Analytical (Linear) compensation chamber temperature.